A STUDY OF THE EFFECT OF AIR DISTRIBUTION ON THE COMBUSTION PROCESS IN A SIMULATED TURBO-JET COMBUSTION CHAMBER

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by

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PREFACE

I would like to take this opportunity to thank the many members of the Mechanical and Aeronautical Engineering Departments who assisted in this investigation and in particular, Professor T. E. Murphy and H. N. McManus for their advice and guidance.

I am also deeply grateful to my wife, Margo, without whose patient forebearance and generous understanding my three years of belated scholastic endeavour could not have been completed.

P. L. A.

May 28, 1953

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INTRODUCTION

A survey of the different components of the present day turbo jet engine shows a considerable volume of data available for the design of all except the combustion chamber. The very nature of the turbo jet places severe requirements on the combustion chamber. These requirements are essentially those of high heat release, low weight, small size, low pressure drop through the chamber, resistance to high temperatures and production of stable combustion at all altitudes and flight velocities.

A great deal of work has been done in the field of combustion, chiefly on the chemistry of combustion, flame front propagation and development of fuels. However this information is of less use to the combustion chamber designer than experimental work of a basic nature on comparable combustion chambers. This sort of experimental work no doubt is going on in industry today, but very little of the data is being published. Their efforts in the main seem to be concentrated more toward a system of trial and error than a systematized analysis and experimental investigation of the fundamental processes. To satisfy the requirements as set forth above rather simple configurations have been developed and are in use in aircraft all over the world. These chambers can produce combustion efficiencies in the neighborhood of 100% at the design condition. However, at other than the design condition the efficiencies drop considerably and maintenance of the flame becomes an increasingly difficult problem.

To accomplish these high heat release requirements in a small volume at high through flow velocities it is necessary to employ a well developed technique of air fuel mixing. This technique must accomplish rapid and complete mixing with the lowest possible pressure loss in a minimum of space. The combustion process from an air introduction view point can be thought of as occurring in two steps. First, the introduction of sufficient air to just initiate combustion and obtain high initial heat release. Second, to provide sufficient additional air to complete combustion, in other words, make up a stoiciometric mixture. For the turbo jet engine these two are not sufficient. The turbine blades rotating at high velocities in these hot gases would be short lived. It becomes necessary then to provide sufficient additional diluent air to cool these combustion products to about 1500°F. The problem becomes more difficult when it is realized that the above must be accomplished in chambers about two feet in length with air flow velocities of 100 feet per second or greater.

Stoiciometric air fuel ratios for hydro-carbons average 15 to 1. The maximum allowable temperatures for the turbine blades require air fuel ratios of about 70 to 1. This air fuel ratio is far too lean to permit combustion unless combustion is first completed with a stoiciometric mixture and then the balance of the air is introduced with a high degree of turbulance to quench the hot combustion gases. The importance of a more complete understanding of the effects of air distribution on the combustion, to the combustion chamber designer, can easily be understood. This

report is concerned with the analysis of the effects of air distribution on combustion in a chamber designed to simulate (as closely as possible) the turbo jet engine operating at about 30,000 feet.

THEORY

The theory of combustion in a turbo jet combustion chamber with its numerous added variables is not too well understood nor has it been thoroughly investigated. In investigating the effects of air distribution on turbo-jet combustion, prime importance must be attached to the delay time or lag in the combustion process.

Delay time is considered as the elapsed time from the input of an element of fuel into the combustion chamber until it has commenced combustion. Delay time may be considered as consisting of two parts: (a) Physical delay time, that time required for a liquid fuel to vaporize and diffuse into the air; (b) A chemical delay time, that time required for the breakdown of complex hydro-carbons and initiation of combustion. Physical delay time appears to be a function of the initial droplet size and the rate of heat transfer. The particular fuel in use and the temperature constitute the primary factors affecting the chemical delay time. Lloyd[2,3] investigated delay time directly by spraying liquid fuel into high temperature air streams (1400 - 1800°F.) and determining the distance to the flame front. If the flow rate were accurately known then the delay time was determined. These experiments determined a relation showing the logarithm of the delay time to be approximately inversely proportional to the absolute temperature of the initial air. It was further shown that there was very little difference in this delay time under the stated conditions for greatly varying types of fuel.

Elliot[4] in interpreting work by Muller on cetene could not develop a linear relationship for log of the delay time versus the reciprocal of the absolute temperature and explained the difference as due to the physical delay time and chemical delay time both obeying the above relation and the sum therefore giving the total delay. The data in the two cases did not match well, Lloyd giving values of 30 millisec at 1500°F. and Muller 30 millisec at 450°F. for two different fuels respectively.

There are many factors which might account for the above, the most likely appearing to be atomization of the fuel. However, Lloyd effectively showed that, while important, degree of atomization is not one of the controlling factors.

One might interpret the above to show that at the high temperatures employed by Lloyd the chemical delay was insignificant, which would indicate that the physical delay times of most hydrocarbons are very similar. Temperature of the initial air appears to be well established as a basic variable in the delay time theory of combustion. Practically all of the work has been done at atmospheric pressures although the effect of pressure on ignition is widely known. Schmidt[5] in work comparing tests in a bomb and in an adiabatic compression apparatus developed the relation:

$$t = \frac{Ae^{B/T}}{p^n}$$

where, t = delay time

e = base, Napierian logarithms

T = absolute temperature

p = pressure

A and B are constants

(The exponent n in the relation above was slightly larger than one.)

This relation was confined strictly to homogeneous gas reactions but may hold for a limited range in a heterogeneous reaction.

The factors effecting delay time appear to be those subject to change with individual unit characteristics. This would be manifest in variations of inlet temperatures, velocities, degrees of mixing, full section velocities, etc. The importance of the delay time factor will be strongest in the primary combustion zone for liquid fuels and to a lesser degree in the secondary zone of gaseous mixture combustion. In combustion initiated with a gaseous type fuel an appreciably shorter delay time could be expected. A shorter length for completion of combustion is in turn predicted.

Since it has been well established that there is an appreciable delay time between the introduction of fuel into the combustion chamber and its combustion, there must likewise be an appreciable distance covered before any single element of the fuel in a steady flow process has been completely burned. It seems logical to assume that this distance is directly a function of the full section velocity as well as the factors affecting delay time.

Combustion of hydrocarbons and air is readily accomplished under ideal conditions with air to fuel ratios of approximately

15 to 1. Some 19,000 BTU/lb are released accompanied by temperatures around 3000°F. The process is dependent upon vaporization of the fuel if in liquid form through mixing with air and introduction of sufficient energy in the form of heat to bring a small part of the mixture to ignition temperature.

If this ideal process is transposed to a cylindrical tube of such dimensions that a given quantity of this fuel air mixture when forced through will attain a given full section velocity then the simplest initial description of a turbo jet combustion chamber has been given. Since velocities vary continuously with time and position it is impossible to defintely speak of one velocity. The use of the term "full section velocity" is merely recourse to an average to be used for discussion and comparison, it being kept in mind that it may not exist. The reference velocity is the velocity computed from the total air mass flow rate, the static pressure and temperature at the combustor inlet, and the maximum cross-section of the combustor with a uniform velocity profile.

Presently available materials for the construction of turbin blades have placed limitations of about 1500°F. on combustion chamber exit mixture temperatures. This limitation is met by employing air-fuel ratios of 70 to 1 or higher, but hydrocarbons will not burn at these lean mixtures. Therefore the air is divided into primary air, sufficient to give approximately stoiciometric mixtures for the combustion, and secondary air or quenching air as it is more popularly known.

A combustion chamber consists essentially of an inner liner or basket and an outer liner or can. The inner basket contains the flame and is perforated in a manner designed to permit passage of the correct amounts of air at the proper positions to accomplish the division and mixing of fuel and air as described. The size and positioning of the holes in the inner basket are critical to the functioning and efficiency of the combustion chamber.

Too much cold quenching air introduced too far upstream can only cause the flame to be extinguished wherever it reduces temperatures below the combustion minimums. This incomplete combustion of the fuel causes a drop in the burner efficiency. It is with this problem of where to commence quenching that this paper is primarily concerned.

It seems reasonable to expect a relation between the length required to complete combustion and the combustion chamber full section velocity. In order for this to be simplified sufficiently for a practical investigation the variables, fuel-air ratio, initial temperature, pressure, etc. must be held constant. We thereby limit ourselves to the variables of air distribution and full section velocity and their effect on the combustion as interpreted by the combustor efficiency.

References:

1. Review of Combustion Phenomena for the Gas Turbin, D. G. Shepherd, ASME Transactions, Vol. 73, Oct. 1951, pp. 921-934.
2. Combustion in the Gas Turbin, P. Lloyd, Proceedings, IME, (London) 1945.
3. The Fuel Problems in Gas Turbins, P. Lloyd, Proceedings, IME, Vol. 159, 1948.
4. Combustion of Diesel Fuels, M. A. Elliot, SAE Quarterly Transactions, Vol. 3, 1949, pp. 490-512.
5. Verbrennungsmotoren, F.A.F. Schmidt, J. Springer, reproduced as FIAT final report number 709, 1946, p. 314.

EQUIPMENT

The Test Unit was designed and built by J. E. Janssen based on an original design by E. T. LaRoe to use an available air supply. The unit consists of two parts, the air and fuel imput ducting and their metering devices and the combustion chamber. Air is supplied by an Allison Aircraft Engine Supercharger driven by a one hundred and seventy-five horsepower tank engine.

The combustor is made of stainless steel, approximately twenty inches in length, five inches in height and two inches in width, see Fig's. 1 and 2. The exhaust end is rectangular in shape and the upstream end circular in the vertical plane. There are 48 separate air ducts with a metering orifice, injection orifice, flow straightener and control damper in each, see Fig. 3. There was a primary air orifice at the center most forward port in the chamber into which the fuel and initial air were injected but this was redesigned as will be described later.

There are 39 chromel vs alumel thermocouples placed symmetrically throughout the chamber to a point 2 inches from the exit, see Fig's. 3 and 4. These are connected to a terminal block by ceramic insulated 20 gage thermocouple wires. Insulated copper wire was used to connect the terminal block to the selecter switch on the test panel outside of the test cell. One chromel-alumel thermocouple is welded to the side plates so that their temperature could be held within safe limits.

Another chromel-alumel thermocouple is located in the fuel inlet duct for obtaining initial fuel temperature. Still another is located

Figure 1

TEST CELL APPARATUS

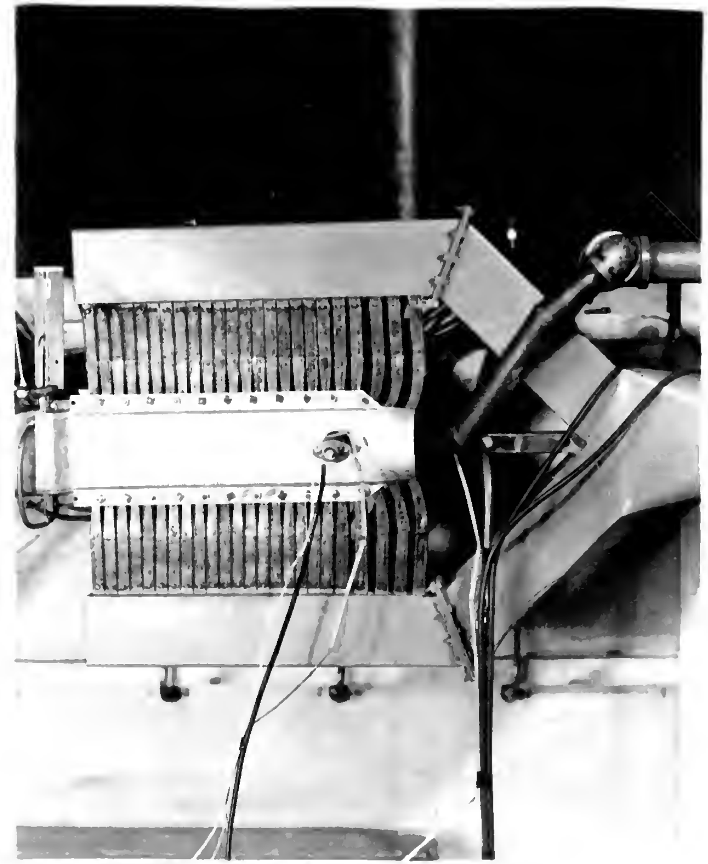


Figure 2

COMBUSTION CHAMBER

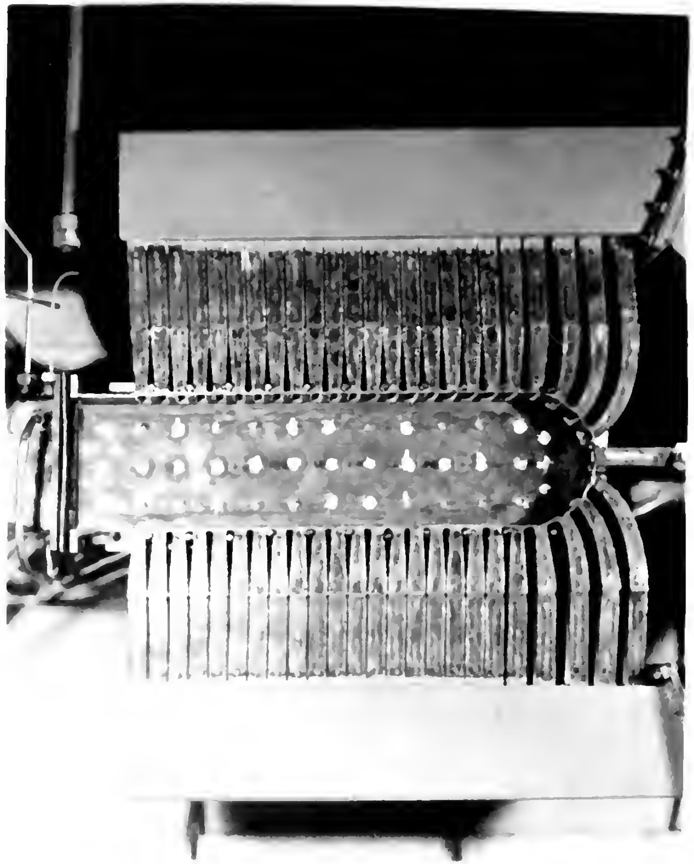


Figure 3

COMBUSTION CHAMBER THERMOCOUPLES
SIDE PLATE REMOVED

Figure 4

ORIGINAL FUEL IMPUT PORT

just downstream of the total flow measuring orifice in the six inch diameter air inlet duct and used to obtain initial air temperature.

In an attempt to more closely simulate an actual turbo-jet combustion chamber and to produce greater mixing the original fuel inlet system was redesigned. Originally the fuel was fed into the primary air duct along with the primary air and then transmitted some five inches into the center of the upstream end of the chamber, see Fig. 4. This design caused a core of air gas mixture less than stoiciometric to enter along the center line at relatively high velocity. To prevent this the primary air duct was closed to all air input and only the fuel was fed into it.

A median fuel flow rate was chosen as the design basis and a fuel splitter designed to divide the fuel into equal parts entering the chamber at an angle of 80° to each other, see Fig's. 5 and 6, a value found to be in common use. The two slots were of such dimensions as to provide an entrance velocity of 100 f.p.s. for the median fuel flow. The design computations are contained in Appendix D.

It was determined necessary to obtain exit velocities and temperatures. A total pressure or impact pressure probe was designed and set into the top of the exit end of the chamber, see Fig. 7. A rubber tube transmitted this pressure to one side of a water filled monometer tube mounted in the panel adjacent to the combustion chamber. Three static pressure take offs were built into the sides of the exit end of the chamber in the same plane with the total

Figure 5

FUEL SPLITTER

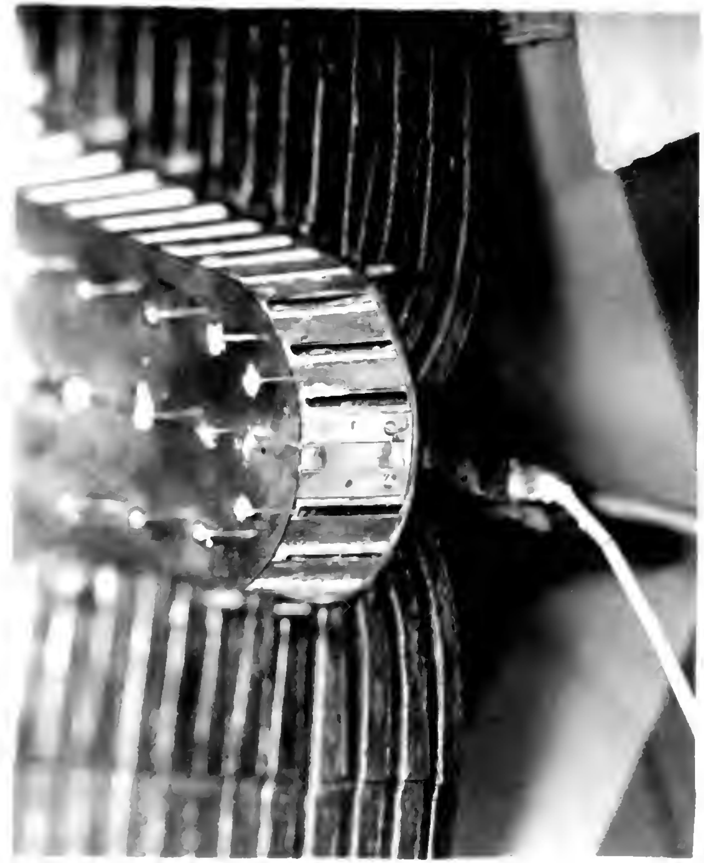


Figure 6

FUEL SPLITTER INSTALLED IN
FUEL IMPUT PORT

pressure probe, see Fig. 9. One was in the bottom center, the second on the left upper half and the third on the right center of the chamber walls. This positioning was considered necessary to as accurately as possible cover the primary flow areas. All three taps were led into a common take off and the averaged static pressure transmitted to the other side of the manometer tube with the total pressure. This arrangement gave ΔP directly for use in computing the velocity.

A temperature probe, see Fig. 8, was designed and installed adjacent to the total pressure probe. It was made of stainless steel tubing into which were fitted ceramic insulators carrying chromel-alumel 20 gauge wire forming a thermocouple. The thermocouple was checked in boiling water, molten tin and an electric furnace up to 1500°F. and found to be accurate to within 2.5°. These leads were brought out to a terminal block where the emf was transmitted to the control panel by insulated copper wire. Both probes were controlled by hand and calibrated for accurate vertical positioning in the exhaust stream. Since alignment of the hole in the total pressure probe relative to the flow direction is critical a guide was added to prevent any possibility of rotation.

A great deal of vibration was encountered in the chamber while operating and errors were introduced into the first of the preliminary runs when the control dampers in the air inlet ports moved. This was corrected by installing friction locks of a high temperature resistant material on each of the dampers. A great deal of leakage was encountered, principally around the control dampers and around

Figure 7

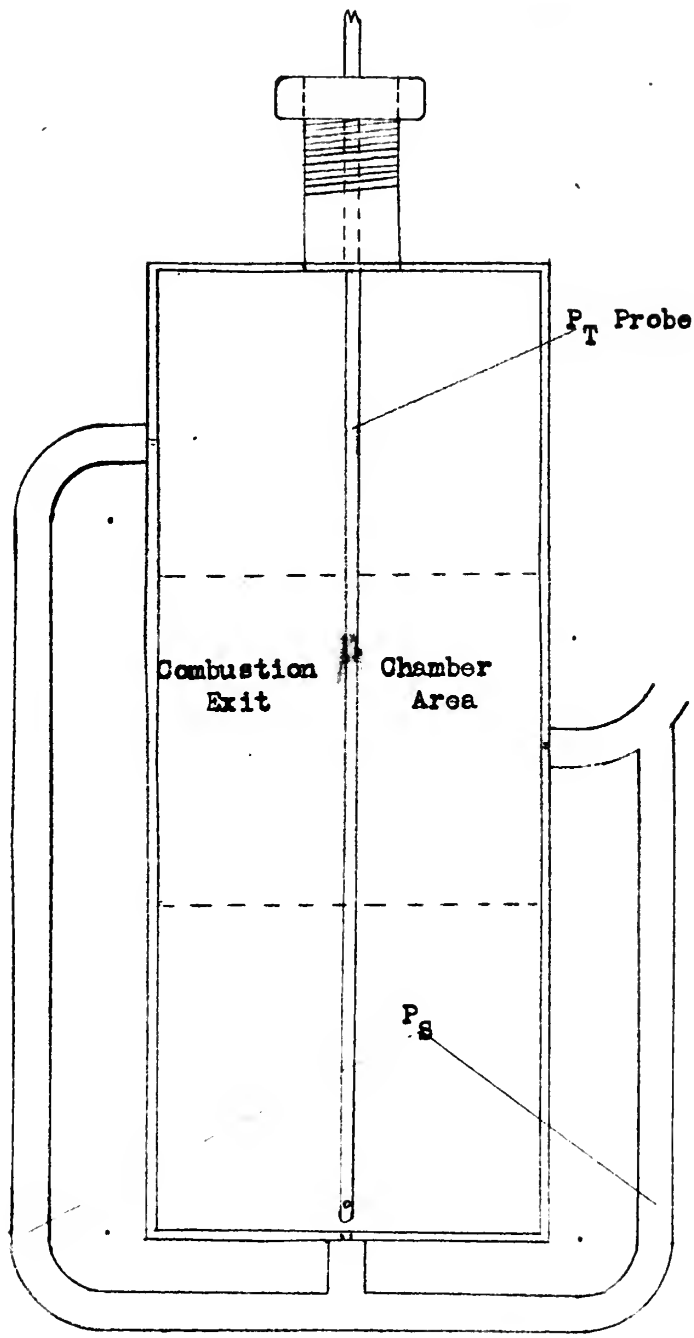
TOTAL PRESSURE PROBE

Figure 8

TEMPERATURE PROBE

Figure 9

STATIC PRESSURE PICKUPS



Scale 1"= 1".

the slip joints. This leakage was substantially reduced by sealing all leaks with a high temperature ceramic cement.

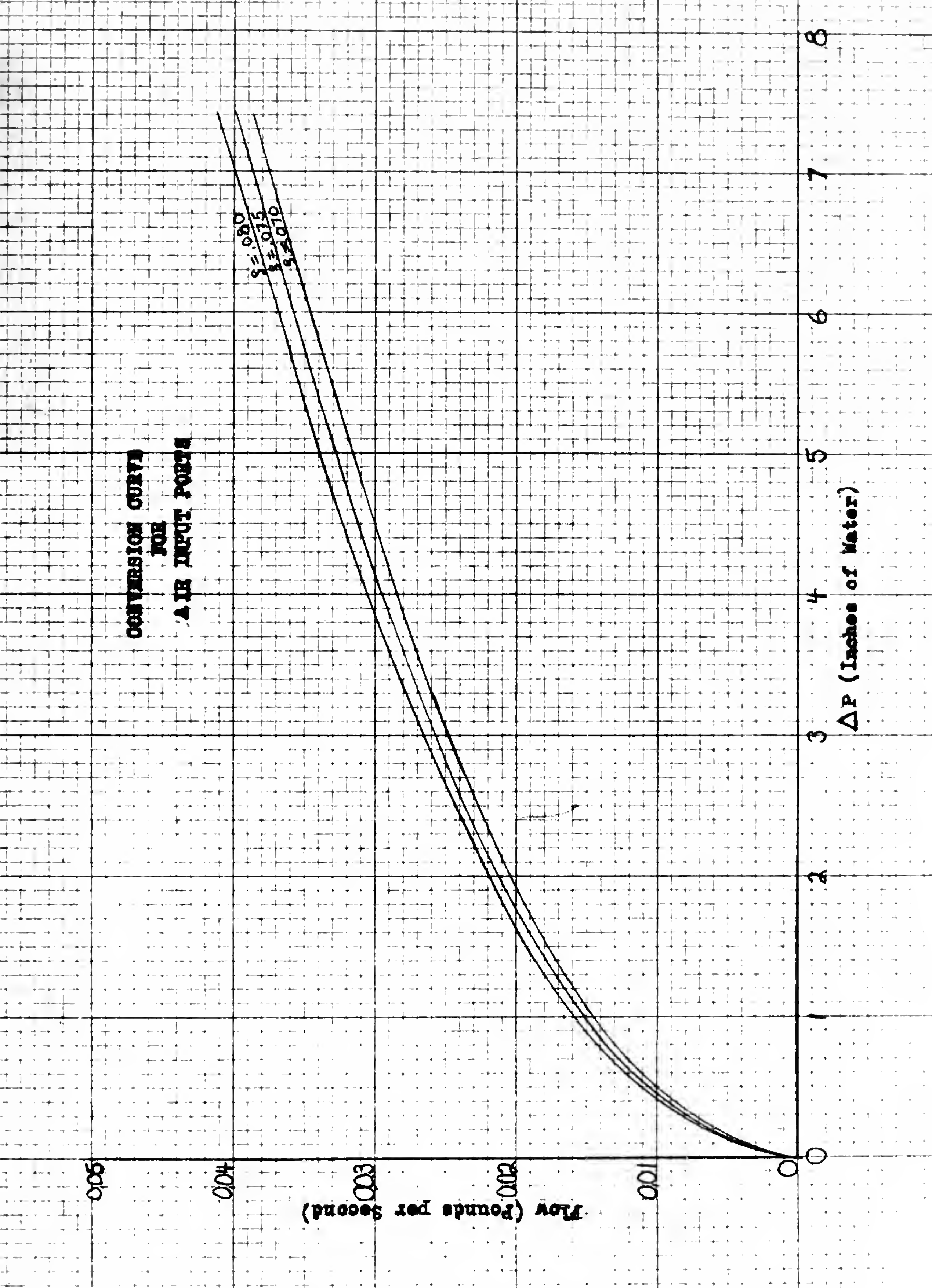
INSTRUMENTATION

Since all the air inlet ports had their own metering orifice and control damper with pressure taps the exact air flow could be accurately controlled. Pressure from each of the pressure taps one upstream and the other downstream of the calibrated metering orifice was carried to a manometer tube bank where the difference in pressure could be measured in inches of water. Using Fig. 10 any predetermined flow of air in lb/sec could be set in each port thus making possible any air input configuration desired.

At the exit end of the chamber the total pressure probe and three averaged static pressure taps were led to a manometer tube mounted on the manometer tube bank. This combination made available, by direct measurement in inches of water, the velocity pressure head to be used in computing the velocity. The total pressure probe had been positioned on the horizontal center line of the chamber and then calibrated to give readings at one-half inch intervals in the vertical plane. This was done to give a series of readings from which the mass flow in each of nine stream tubes could be calculated.

The temperature probe was mounted in exactly the same manner as the pressure probe but due to space limitations it had to be three quarters of an inch farther downstream. The error introduced by this positioning was considered negligible in magnitude and common to all runs. The temperature probe was calibrated so that its readings were taken from the center line of the stream tubes set up

Figure 10


CONVERSION CURVE
FOR
AIR INPUT PORTS
ρ=.080
ρ=.075
ρ=.070
ΔP (Inches of Water)
Flow (Pounds per Second)
0
1
2
3
4
5
6
7
8
.001
.002
.003
.004
.005

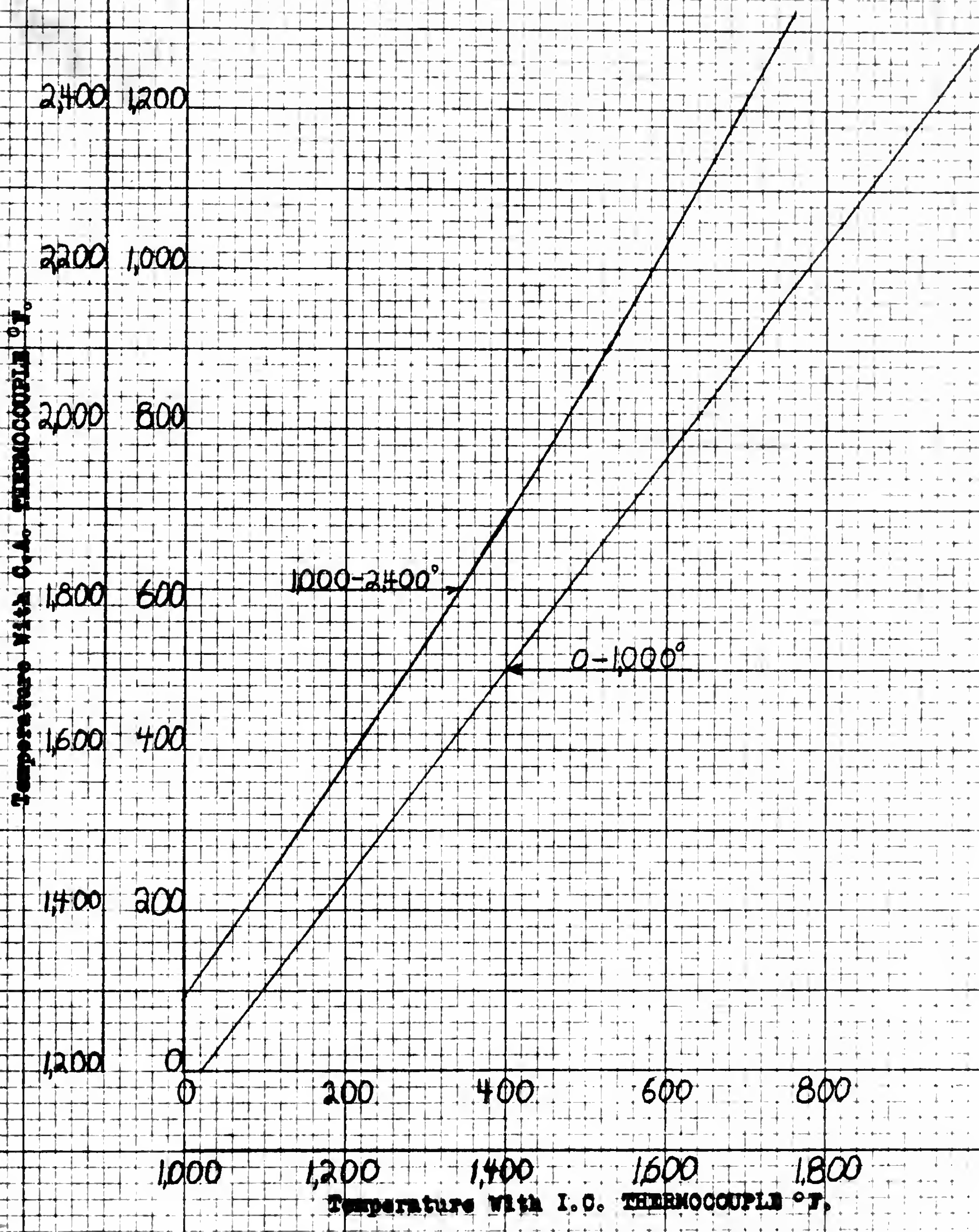
by the pressure probe calibration. This gave in each instance data that matched by position for each run. The EMF was picked up at the control panel and temperatures were read directly from potentiometers. These values were then corrected in each case by Fig. 11.

The thermocouples mounted permanently within the combustion chamber were read by potentiometers and their values corrected from Fig. 11, since all thermocouples were of the chromel-alumel type. The Butane was obtained from 100 lb. bottles thru a constant pressure valve set at 5 psi. The flow was metered by using a needle valve and measured with a Fischer and Porter flowrator Model 5A-25 calibrated in C.F.M. of air at standard conditions of 14.7 psia and 70°F. A correction was applied to the meter reading as shown in Appendix A.

The usual engine control instruments, RPM, oil temperature, oil pressure and throttle were mounted on the control panel. In addition oil temperature and pressure gages for the compressor were mounted on the panel and since the compressor-engine gear ratio was 10 to 1 the compressor RPM was known.

Figure 11

THERMOCOUPLE CONVERSION CURVE



ESTIMATED ACCURACY OF MEASUREMENTS

To a great extent the accuracy of the measurements are dependent upon the estimates made by Janssen,[1] the designer and builder of the equipment. These estimates include a plus or minus 3% from the calibration curves of Fig. 10 used for the secondary-air-metering orifices. The flowrator employed to measure the fuel flow is of standard make and essentially an orifice meter of estimated accuracy within a plus or minus 2%.

The temperatures recorded by the many chromel-alumel thermocouples are corrected by Fig. 11 for variation from iron-constantine correction in the potentiometers. Fluctuations of 15°F. were encountered at temperatures from 500 - 1500°F. and were most noticeable in the probe traversing of the chamber exit. Considerable error could be expected due to the fact that measurements with this type of thermocouple are usually some value between T_{total} and T_{static}.[2] They are also subject to radiation from the flame but this can be neglected since the thermocouple is quite small. The heat then is transferred to the junction largely by convection and away from the junction principally by radiation to the chamber walls. The maximum temperatures recorded by the probe had a radiation error of 25° which is within 3% and was omitted in the computations.

Pressure measurements are considered accurate to 1/20 of an inch of water. This variation is the result of the turbulent flow in the chamber and the corresponding pressure fluctuations which

caused the water levels in the manometer tubes to vary continuously. Consequently the values measured are averaged values. In the preliminary runs employed to determine "bugs" in the equipment and to standardize operating procedure, it was first attempted to pre-set a weight flow of air at the primary metering orifice and then distribute this in a predetermined manner thru the combustion chamber ports. This was proved not feasible for several reasons. At the lower weight flows the engine was forced to idle to hold down compressor RPM and as a consequence ran too roughly for acceptable tolerances to be maintained. There was an appreciable air loss due to leakage in the slip joints and around the metering valves between the primary orifice and the individual port orifices which was not too accurately known.

The primary power plant was found to operate within a plus or minus 10 RPM at 1000 RPM and this setting was employed throughout. This required throttling at each of the combustion chamber input ports but gave steady readings when the tendency of the valves to vibrate out of position was corrected. Losses in the ducting were then of no importance.

References:

1. A Preliminary Investigation into the Effect of Air Distribution on Mixing in a Constant-Pressure Combustion Chamber, J. E. Janssen, an M.S. Thesis submitted to the University of Minn. Jan. 1953.
2. Measurement of High Temperatures in High Velocity Gas Streams, W. J. King, ASME Transactions, Vol. 65, 1943.

INVESTIGATION PROCEDURE

The variables entering the test process were prohibitive in number. It was determined therefore to set and maintain one air-fuel ratio, to investigate three full section velocities, to retain one initial air fuel ratio of 20 to 1, to introduce the initial air thru 10 ports, 5 on top and 5 on the bottom of the burner, and to introduce the quenching air thru 16 ports, eight on top and eight on the bottom.

The critical distance "L" would be that from the fuel input point to the first of the quenching ports. All ports between those conducting primary air for combustion and those conducting quenching air would be set to bleed a minimum of air to prevent burning back into the port. The bank of eight quenching ports on top and eight on the bottom would be moved as a unit to vary the volume allowed for combustion. Their travel would be limited by the forward point of no combustion and the exit of the chamber.

Sample preliminary calculations employing averaged values determined by the designer and other values chosen for the particular step in the investigation are shown in Appendix A. The investigation was broken into three steps, one each for full section velocities of 75, 100 and 125 fps. Initially an air-fuel ratio of 70 - 1 was employed in the preliminary runs but this developed temperatures within the combustion chamber in excess of the maximum allowable for the chromel-alumel thermocouples. The final air-fuel ratio used throughout the investigation was 120 - 1.

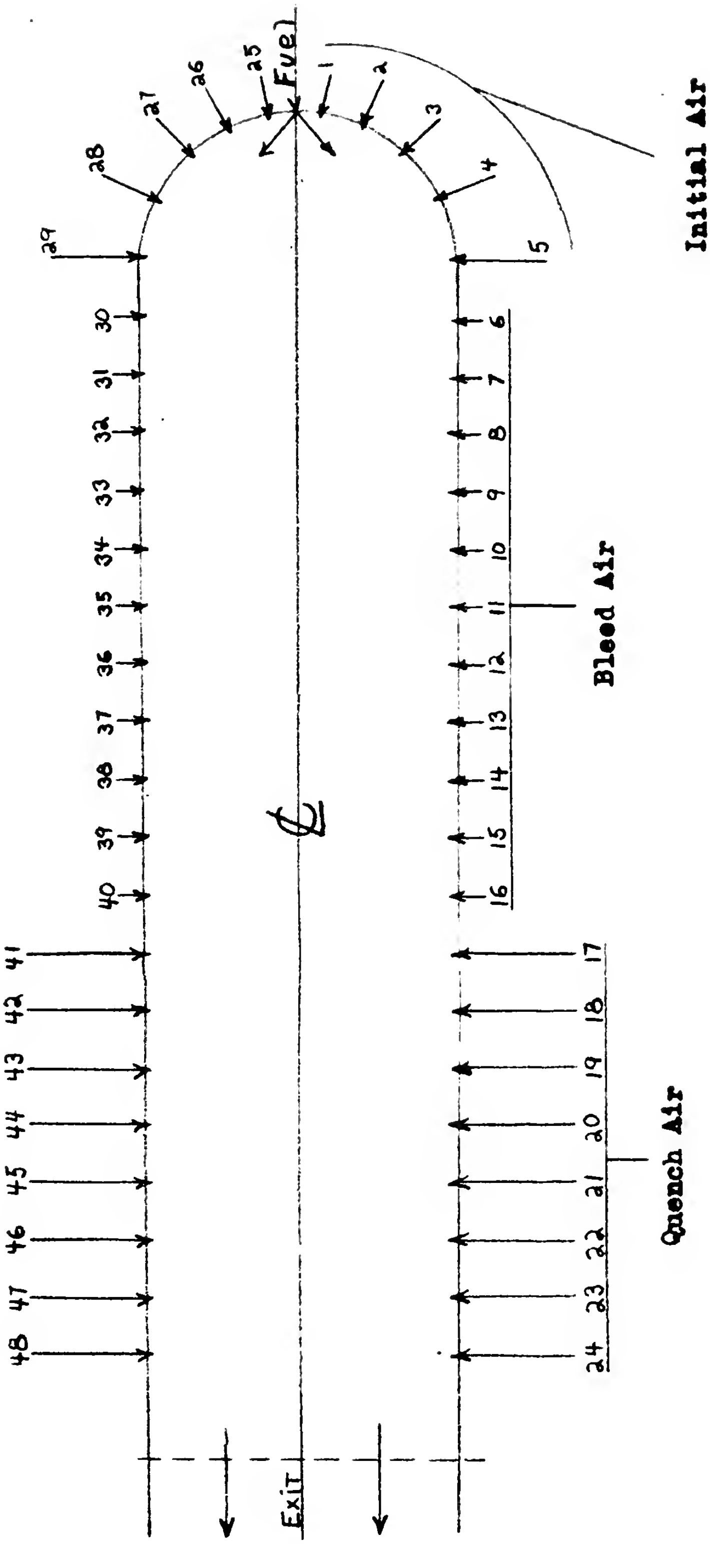
Each full section velocity determines a required weight flow of air. This flow is divided in a manner to provide a primary air-fuel ratio of 20 to 1, a portion of the balance for bleed air to prevent burning into the ports and for cooling and the balance is employed as the quenching air, see Fig. 12. The ports are numbered consecutively from entrance to exit, 1 to 24 on the bottom and 25 to 48 on the top.

The engine driving the air compressor is started in the usual manner and allowed to reach operating temperature. One thousand RPM is then set and maintained by an operator at the test panel.

Without combustion the entire predetermined weight flow of air, primary, bleed and quenching is set while the total flow is held constant. This is done by presetting in the manometer tube bank the required value of ΔP in inches of water for each port. Since any change in one port will affect slightly the values in the others, this presetting requires at least three complete checks and adjustments for all ports. With this completed the spark is started and the predetermined weight flow of fuel is injected. Combustion should commence within 30 seconds. A minimum of three operators are considered necessary to safely and accurately operate the equipment.

Combustion is allowed to continue for five minutes to permit all components to reach operating temperatures. While one operator holds the RPM and fuel flows constant, a second traverses the exit cross section temperature probe and the third records the readings from the potentiometer. While two operators are simultaneously taking down the chamber temperatures the third is measuring the

Figure 12



SCHEMATIC, COMBUSTION CHAMBER FLOWS

values of ΔP for the exit cross section with the pressure probe. Initial air and fuel temperatures are the final readings obtained.

The probe temperatures as read from the potentiometers are corrected by Fig. 11 for chromel-alumel to iron constantine. This corrected temperature is then employed in Fig. 13 with the average static pressure to determine the density in per cent of standard density. With per cent ρ and ΔP, Fig. 14 gives velocity in fps. The mass flow weighted temperature rise ΔT_b is obtained as shown in Appendix B. The equation for burner efficiency as employed herein is:

$$\eta_b = \frac{\text{actual entholpy rise}}{\text{ideal entholpy rise}}$$

$$= \frac{T_{avg} \text{ X } C_{p\ avg}}{19{,}670 \text{ X } F/a}$$

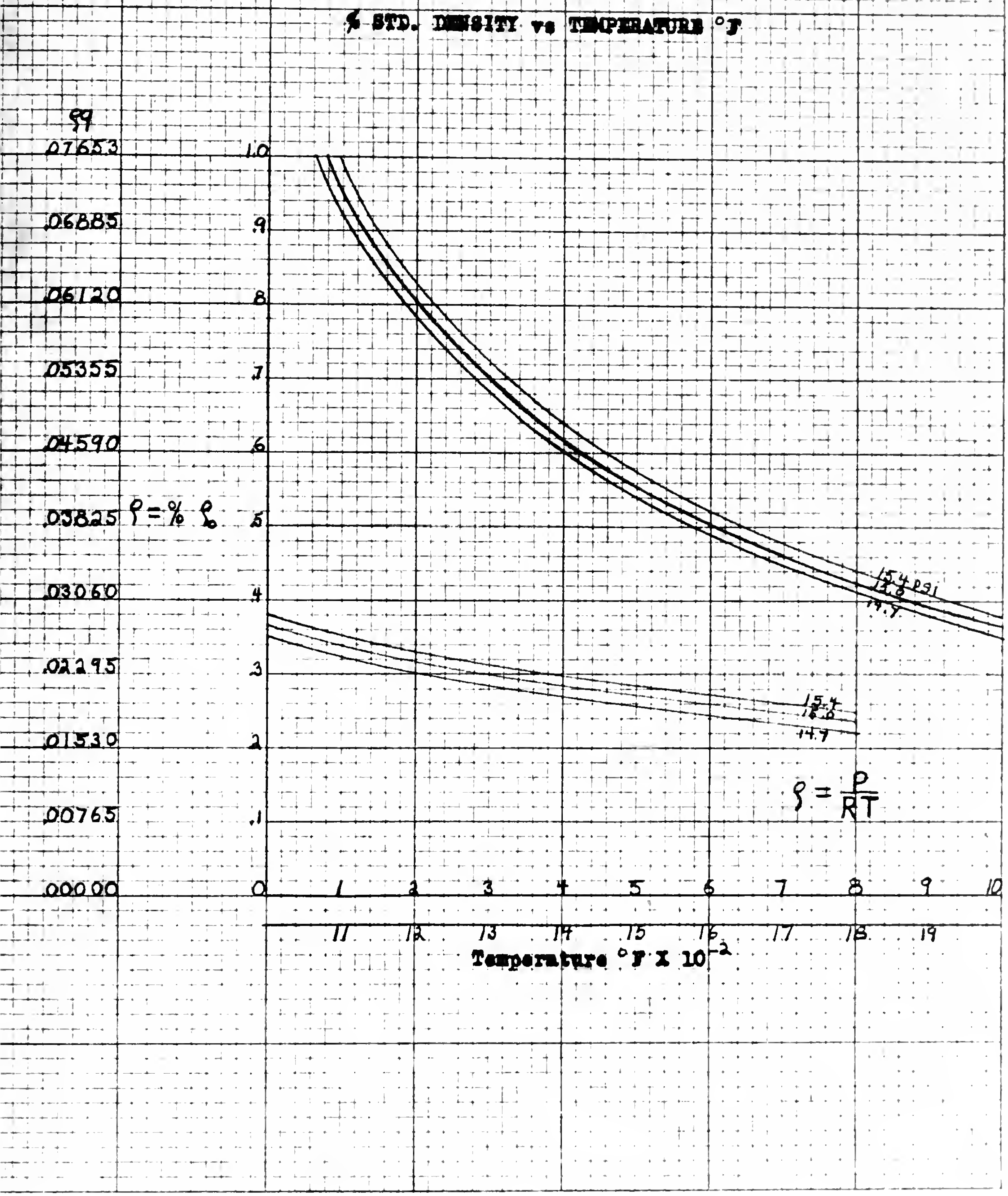
Where η_b = burner efficiency

$\Delta T_{b\ avg}$ = mass weighted average temperature rise across the burner

$C_{p\ avg}$ = average value for range of temperatures employed

F/a = fuel-air ratio

Figure 13


% STD. DENSITY vs TEMPERATURE °F
ρρ
.07653
.06885
.06120
.05355
.04590
.03825
.03060
.02295
.01530
.00765
.00000
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
0
ρ = % ρ₀
15.4 psi
15.0
14.7
15.4
15.0
14.7
ρ = P/RT
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
Temperature °F X 10⁻²

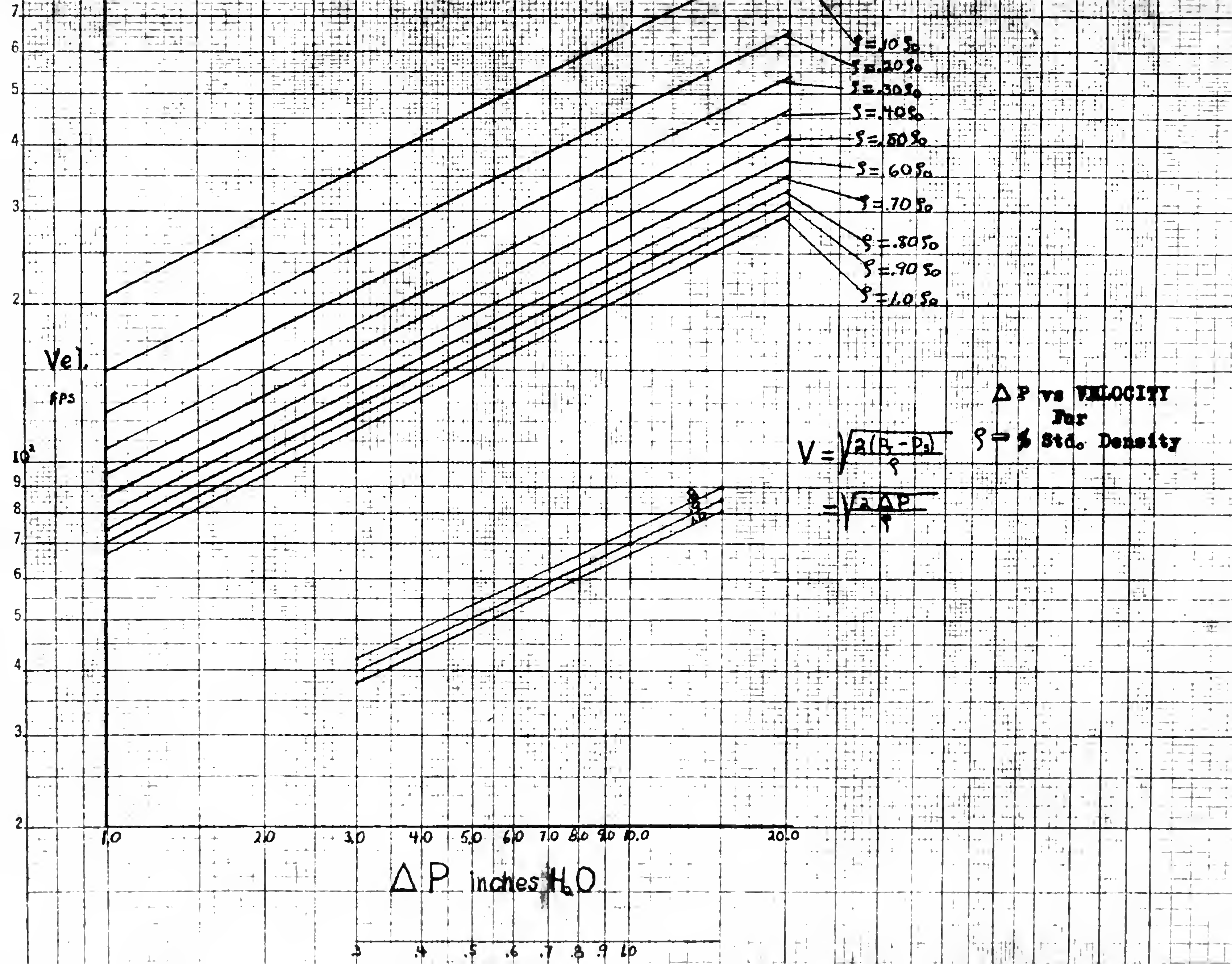
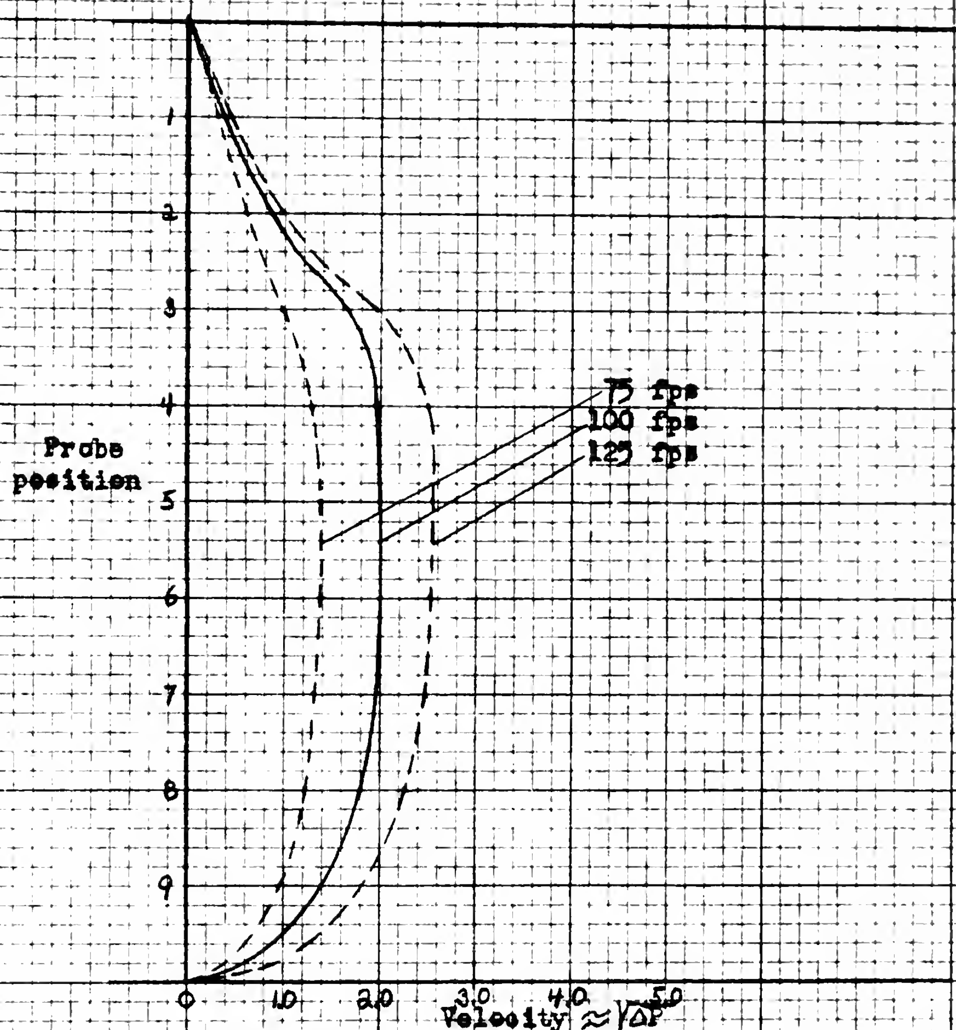

ΔP vs VELOCITY
For
ρ = % Std. Density
$V = \sqrt{\frac{2(P_T - P_S)}{\rho}}$
$= \sqrt{\frac{2\Delta P}{\rho}}$
ρ = .10 ρo
ρ = .20 ρo
ρ = .30 ρo
ρ = .40 ρo
ρ = .50 ρo
ρ = .60 ρo
ρ = .70 ρo
ρ = .80 ρo
ρ = .90 ρo
ρ = 1.0 ρo
Vel.
FPS
ΔP inches H2O
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
20.0
.3
.4
.5
.6
.7
.8
.9
1.0

Figure 14

Figure 15

VELOCITY PROFILES
Without Combustion



throughout. A definite drop, in the temperatures across the exit, was shown to occur when quenching was attempted too far upstream as was expected from theory.

An incident of burning back into a closed port with flames escaping thru an expansion joint occurred. This was corrected by maintaining a positive pressure of .1 inch of water in all "closed" ports. A condition of unbalance and unacceptable pressure fluctuation was found to exist when the compressor was operated at low RPM to permit the total predetermined weight flow of air to be metered by the total flow metering orifice. Metering at this orifice was omitted and engine speed held constant at 1,000 RPM. This permitted attainment of sufficient pressure and weight flow for accurate and steady throttling of the flow into each port.

As the block of quenching ports reached the most downstream position an attempt was made to reduce their number from the original eight on top and bottom to seven on top and bottom, etc. This required an increase in the weight flow and the velocity for each port. A decrease in the burner efficiency was obtained under these compacted quenching conditions. This can be explained by a greater penetration and mixing of the quenching air than had been obtained under the initial configuration causing a lower average value for the chamber temperature rise. This compacting of the quenching air was not employed in the final runs.

DISCUSSION OF STEPS I, II AND III

Steps I, II and III are a series of runs at 75 fps, 100 fps and 125 fps, respectively. A total of 43 runs were involved in the work and the data from the final runs for each step is tabulated in Figure 16.

Step I required a series of 9 runs commencing at the exit end of the combustion chamber and moving one port forward in each run. The combustion in all runs was smooth with no tendency towards pulsating roughness until port No. 9 was reached. With the quenching-air block of ports commencing with No. 9 the combustion was rough and unsteady and had to be restarted five times before completion. Combustion with Port No. 8 as the first of the quench ports was found to be impossible even though combustion could be started at reduced velocities it would blow out before the required full section velocity of 75 fps was reached.

Figure 17 shows a plot of the Burner Efficiency for each run versus the length L in inches to the first quenching port. It is of interest to note that there is a definite minimum length, for the configuration employed, below which it is impossible to obtain combustion. Correspondingly there is another value of L beyond which the rate of change of efficiency with combustion length is practically zero. In Step I at the most forward point of quenching, there were only three inches between the last port admitting initial-air and the first port admitting quenching-air.

Figure 16

TABULATION OF DATA

STEP I

Vel_{fs} = 75 fps A/F = 120 Fuel = Butane W_a = .398 lb/sec W_F = .00328 lb/sec

Run #	L"	T_F °F	ΔT_b avg.	η_b%	T_1 °F	V_e avg. Ft/sec	P_{es} psia	Comb.
1	6.28	70	487	74.10	90	155.0	14.75	rough
2	7.00	60	513	78.38	80	150.0	"	smooth
3	7.75	62	514	78.40	80	157.0	"	"
4	8.50	62	504	77.00	79	162.0	"	"
5	9.25	65	520	79.40	88	161.0	"	"
6	10.00	70	511	78.00	90	162.0	"	"
7	10.75	70	535	81.50	90	162.0	"	"
8	11.50	70	512	78.10	80	163.0	"	"
9	12.25	70	514	78.40	80	165.0	"	"

STEP II

Vel_{fs} = 100 fps A/F = 120 Fuel = Butane W_a = .532 lb/sec W_F = .00445 lb/sec

Run #	L"	T_F °F	ΔT_b avg.	η_b%	T_1 °F	V_e avg. Ft/sec	P_{es} psia	Comb.
1	7.75	70	352	53.00	80	190.0	14.79	rough
2	8.50	75	387	58.50	80	184.0	"	smooth
3	9.25	81	400	60.50	80	192.0	"	"
4	10.00	82	500	76.40	82	196.0	"	"
5	10.75	82	520	82.50	82	201.0	"	"
6	11.50	85	555	85.10	85	198.0	"	"
7	12.25	90	548	84.00	90	196.0		"

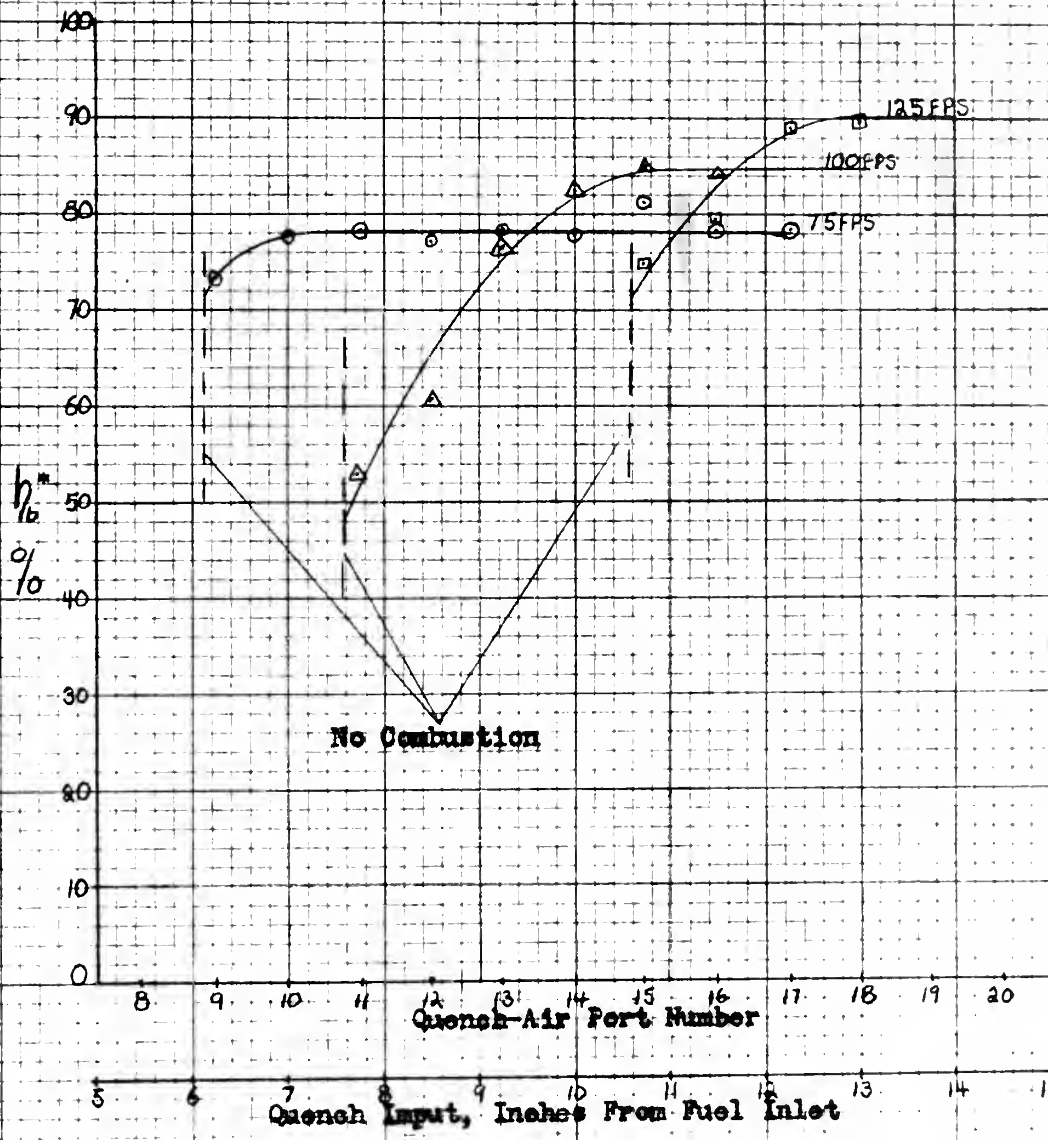
STEP III

Vel_{fs} = 125 fps A/F = 120 Fuel = Butane W_a = .663 lb/sec W_F = .00552 lb/sec

Run #	L"	T_F °F	ΔT_b avg.	η_b%	T_1 °F	V_e avg. Ft/sec	P_{es} psia	Comb.
1	10.75	81	487.5	75.0	110	231.0	14.82	very rough
2	11.50	81	510.0	78.5	110	234.0	"	rough
3	12.25	81	573.0	88.3	110	258.0	"	smooth
4	13.00	81	577.0	89.2	110	253.0	"	"

Figure 17

BURNER EFFICIENCY vs COMBUSTION LENGTH
a/f= 120



*These are relative efficiencies.

Step No. II employed a full section velocity of 100 fps and the burner efficiency for each run is plotted versus L in Figure 17. The quenching was commenced at the exit of the chamber and moved forward a total of seven ports. Combustion was smooth and steady until port No. 11 was reached, where it became extremely rough and pulsating and had to be restarted three times. Combustion with port No. 10 as the first of the quenching ports was found to be impossible. The curve in Figure 17 for 100 fps is markedly similar to that obtained in Step No. I except that the maximum burner efficiency is roughly 5% greater. The last part of the curve which was impossible to obtain experimentally has been extrapolated on the basis of the curve obtained in Step No. I.

It is of interest to note that the efficiency dropped steadily over a length of approximately four inches and five ports before a point of no combustion was reached. This is in marked contrast to the case in Step No. I.

Step No. III involved a full section velocity of 125 fps and is plotted in Figure 17. A curve similar in all respects to that obtained in Step No. II was obtained except that the maximum efficiency is again about five per cent higher than the previous step. A total of four runs were obtained and the latter part of the curve is extrapolated in a manner conforming to the curve of Step No. I. Combustion was extremely rough and appeared to have cyclic pulsations which would continue until the flame was extinguished. As the points of maximum efficiency were reached however, the combustion stabilized and appeared normal in all respects. The pulsation could be caused by the flames

gradually being blown downstream with a system of flare ups occurring in the process. When the beginning flame front had been moved downstream to the vicinity of the first quench ports the balance of heat transferred to the unburned fuel air mixture steadily decreased until combustion stopped.

Sketches of the probable combustion area within the combustion chamber for each of the three steps are shown in Figures 18, 19 and 20. The sketches are based on the temperatures within the combustion chamber and a temperature of 1400°F. was employed as the minimum for combustion. The three runs chosen were those at the first points of maximum efficiency. Since the thermocouples within the combustion chamber were 1.5 inches apart, the distance to the probable flame front was extrapolated.

Figure 21 is a plot of combustion length versus full section velocity with parameter of first point of maximum efficiency. This curve on first thought would appear to have a point of zero velocity for zero length. This is hardly feasible when it is realized that for extremely small velocities and low flow rates the latter ports of the initial air configuration would be in a position similar to the quenching ports. On this basis it is believed that the curve would have no meaning if carried forward of the last initial air port, in this case a point approximately 3.5 inches from the fuel inlet.

The leveling out of the curve at the higher velocities indicates a capability of the burner to handle higher velocities at a much reduced rate of increased burner length.

Figure 18

PROBABLE COMBUSTION AREA

Velocity$_{ts}$ = 75fps; Temperatures in°F
Minimum Combustion Temperature = 1400°F

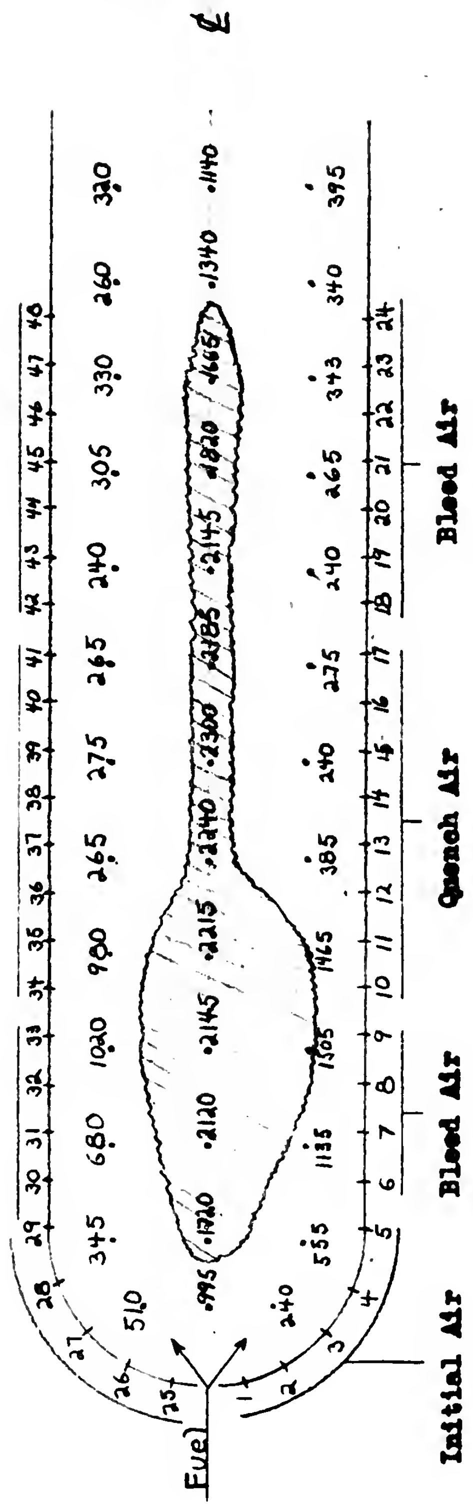


Figure 19

PROBABLE COMBUSTION AREA

Velocity = 100fps; Temperatures in °F
Minimum Combustion Temperature = 1400°F

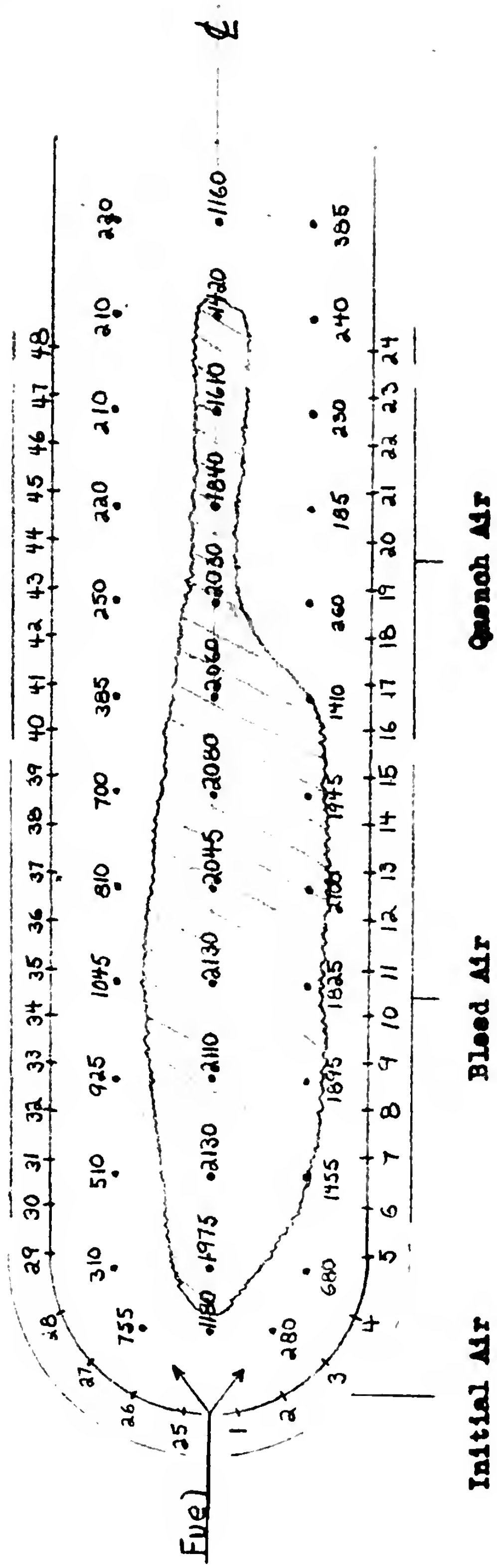


Figure 20

PROBABLE COMBUSTION AREA

Velocity = 125 fps; Temperatures in °F
Minimum Combustion Temperature = 1400°F

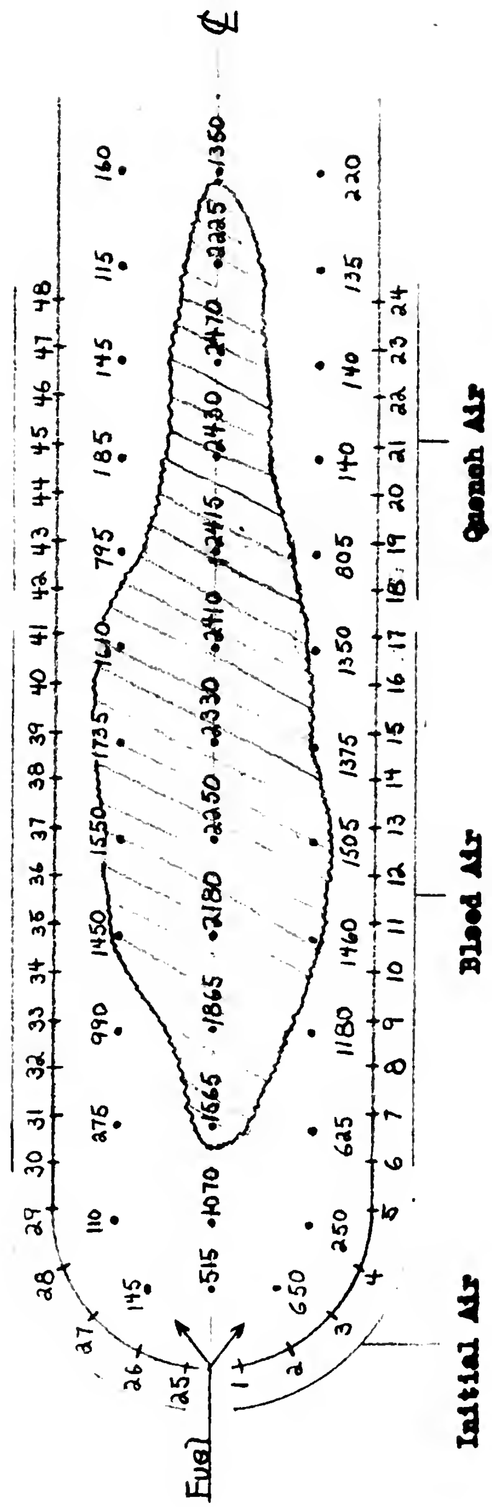
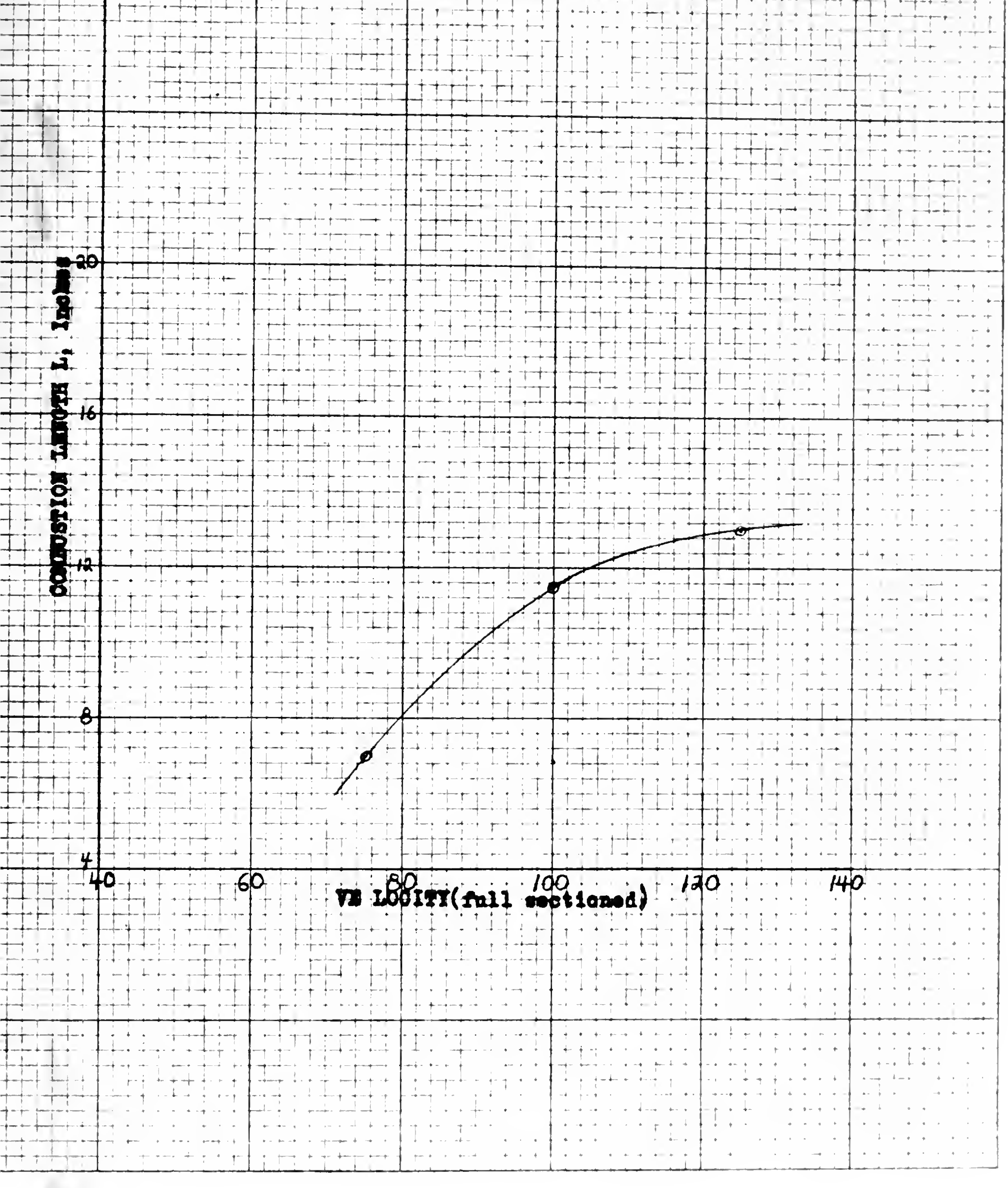


Figure 21

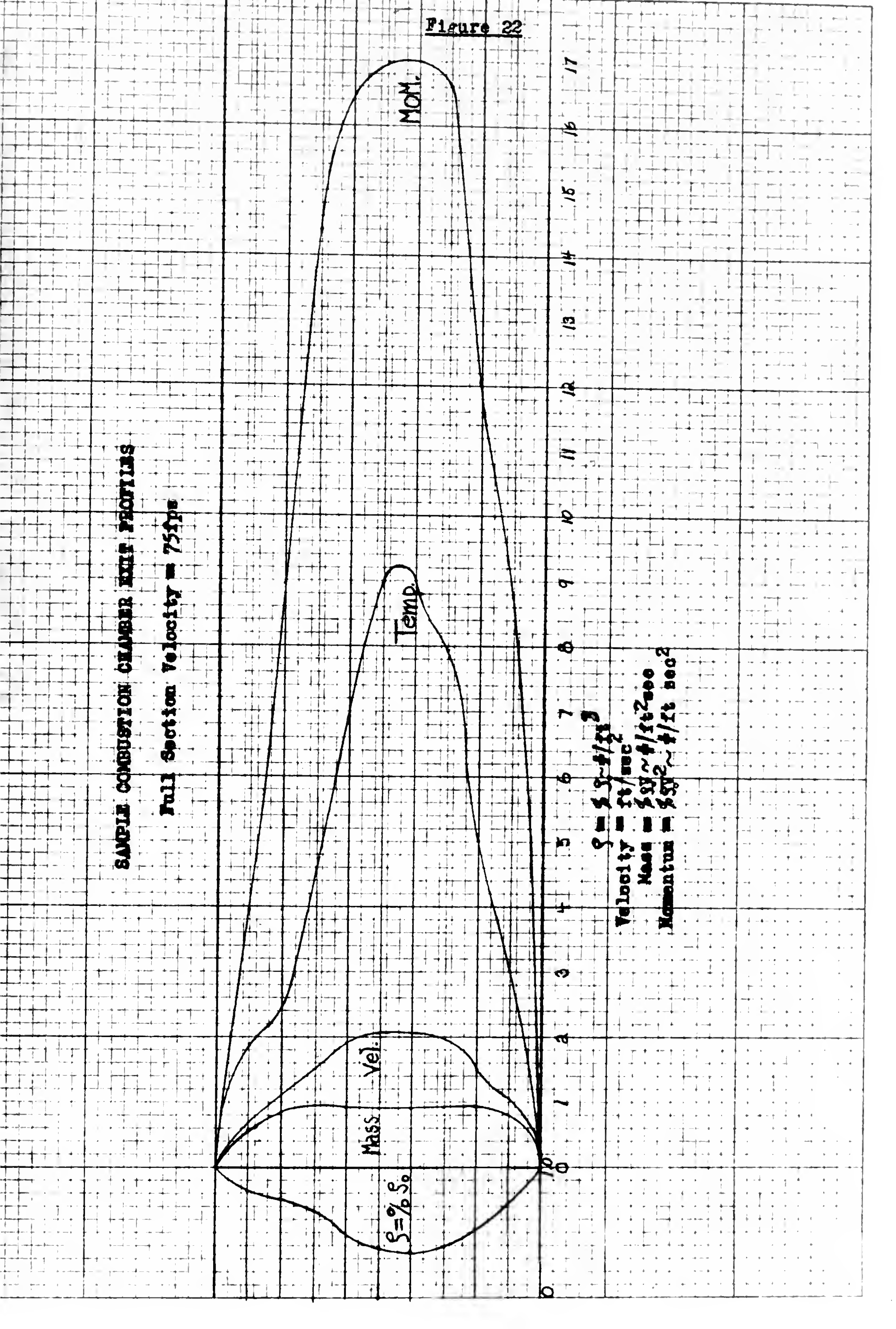
COMBUSTION LENGTH vs VELOCITY$_{fs}$
FOR
η_b =Max. ; L = Min. ; a/f =120



An extremely poor temperature profile for the exit cross section was obtained in all runs. No attempt was made to correct this profile after the asymmetrical velocity profile was found to be uncorrectable. Figure 22 is a plot, for a representative run, of the density, velocity, temperature, mass flow and momentum profiles at the chamber exit. Scales have been proportioned, except for the density, to give a representative indication of magnitude.

The increase in maximum efficiency between each step approximates 5%. This is in a sense difficult to explain since in each step the air-fuel ratios are the same, initial air temperatures vary within a range of 40°, mass flows of air are equally proportioned and similarly placed, all recording and computational methods are the same, and in each case combustion is completed as shown by the maximum temperatures recorded in the exit traverses. There remains the factor of degree of mixing which can logically be expected to increase with imput velocities to the chamber. A comparison of the probable combustion areas as shown in Figures 18, 19 and 20 shows a progressive broadening as well as lengthening of the areas with increased full section velocity. This indicates a decreasing volume along the top and the bottom of the burner, in which temperatures are below the combustion minimums, thru which fuel could pass and never become ignited. This same condition can be presumed to exist in the horizontal as well as the vertical plane due to side wall quenching effect.

Figure 22


SAMPLE COMBUSTION CHAMBER EXIT PROFILES
Full Section Velocity = 75fps
ρ = % ρo ~ #/ft³
Velocity = ft/sec
Mass = %ρV ~ #/ft² sec
Momentum = %ρV² ~ #/ft sec²
MOM.
Temp.
Vel.
Mass
ρ=%ρo
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17

CONCLUSIONS AND RECOMMENDATIONS

From the data recorded and subject to the conditions and restrictions as imposed heretofore the following appear to be of general interest to the combustion chamber designer.

a. In a constant pressure, steady flow type of combustion chamber at one air-fuel ratio there is a definite minimum length required for completion of combustion within which quenching air will stop the combustion.

b. There is likewise a maximum length, dependent upon the full section velocity, for completion of combustion beyond which the burner efficiency is practically constant.

c. There is a tendency at least within the range of full section velocities covered herein for the burner efficiency to increase with increased velocity, other factors remaining the same.

It is recommended that a similar series of tests be run employing liquid fuels to determine variations if any in combustion lengths for gaseous and liquid type fuels.

A series of runs employing variable air-fuel ratios for constant full section velocities would show this variable's effect on combustion length. The equipment is considered capable of producing a great deal of relative information of value to the study of combustion problems in turbo jet type combustion chambers.

APPENDIX

APPENDIX A

SAMPLE PRELIMINARY CALCULATIONS

Preliminary calculations were required for each step in the investigation, i.e. for 75, 100 and 125 fps full section velocity. The values required were:

a. Total air flow in lb/sec.

b. Total fuel flow in lb/sec, of Butane.

c. Corrected fuel flow rotometer setting.

d. Initial air flow in lb/sec port.

e. ΔP in inches of water for each port in initial air configuration.

f. Total air flow required for bleed air in lb/sec.

g. Total air flow required for quenching air in lb/sec.

h. ΔP in inches of water for each port in the quench air configurations.

Assume:	Full Section Velocity	= 100 fps
	Air - fuel ratio	= 120 by wt.
	Chamber Exit Area[1]	= .0695 ft^2
	Chamber Static Pressure[1]	= 14.7 psi
	Specific Density Butane	= .164 lb/ft^3
	Specific Density of Air	= .0765 lb/ft^3

Total Air Flow = 100 fps X .0695 = 6.95 ft^3/sec = 31.95 lb/min = .532 lb/sec

let w_f = wt. flow of fuel for air-fuel ratio of 120 then

$$w_f + 120\, w_f = 31.95 \text{ lb/min}$$

$$w_f = .264 \text{ lb/min} = .00445 \text{ lb/sec}$$

$$w_f = \frac{.264 \text{ lb/min}}{.164 \text{ lb/ft}^3} = 1.61 \text{ ft}^3\text{/min}$$

The fuel flow must be corrected to the standards of the rotometer. The correction factor of .77 is obtained from the nomograph on page 9823 of the Fischer and Porter Company catalogue, section 98 - A, entitled "Theory of the Flowrator." This is based on average values of

$$P_f = 15.3 \text{ psia}$$

$$T_f = 525^\circ R$$

$$\text{Sp. Gravity} = 2.08$$

Therefore rotometer fuel setting $= \frac{1.61}{.77} = 2.095 \text{ ft}^3\text{/min.}$

Assuming an initial air flow based on an air-fuel ratio of 20/1:

$$20/1 = W_{ai}/.264$$

$$W_{ai} = 5.3 \text{ lb/min} = .0884 \text{ lb/sec}$$

This weight flow of air is to be introduced through ten ports, five on top and five on the bottom of the combustion chamber. The following arbitrary per cent for each port was determined during the preliminary runs.

Ports No. 1 and 25	7.5% each	= .00675 lb/sec port
Ports No. 2 and 26	9.5% each	= .00865 " " "
Ports No. 3 and 27	9.5% each	= .00865 " " "
Ports No. 4 and 28	11.0% each	= .00975 " " "
Ports No. 5 and 29	12.5% each	= .01040 " " "

From Fig. 10 ΔP for ports No. 1 and 25 = .30" H_2O

" " "	2	"	26	= .38" "
" " "	3	"	27	= .38" "
" " "	4	"	28	= .44" "
" " "	5	"	29	= .51" "

In all configurations there are 22 ports, 11 on top and 11 on bottom, passing bleed air at $\Delta P = .10'' \ H_2O = .00436$ lb/sec. Total bleed air,

$$W_{ab} = 22 \times .00436 \text{ lb/sec} = 5.76 \text{ lb/min}$$

Therefore total imput of air up to first quenching ports,

$$W_{aib} = 5.3 + 5.76 = 11.06 \text{ lb/min.}$$

This leaves a balance for quench air

$$W_{aq} = 31.95 - 11.06 = 20.89 \text{ lb/min} = .3485 \text{ lb/sec}$$

This weight flow is put in through 16 ports, 8 on top and 8 on the bottom.

$$W_{aq/p} = .3485/16 = .0218 \text{ lb/sec port}$$

$$\Delta P \text{ for quenching} = 2.1'' \ H_2O$$

References:

1. A Preliminary Investigation Into the Effects of Air Distribution on Mixing in a Constant-Pressure Combustion Chamber, J. E. Janssen, an M. S. Thesis submitted to the University of Minn. Jan., 1953.

APPENDIX B

COMPUTATION OF AVERAGE CHAMBER TEMPERATURE RISE

The average temperature rise for the combustion chamber was obtained by weighting the probe temperature as follows. The vertical dimension of five inches was divided into ten equally spaced segments forming for the flow a number of equal streamtubes. Temperatures were then obtained at the center of each streamtube and the value of density for the original static pressure obtaining was read from Figure 13. ΔP was obtained with the total pressure probe and static pressure leads at the boundaries of each streamtube and then averaged. The velocity was read from Figure 14 for each set of values of ΔP and ρ.

Let X_i = density in % of standard for each streamtube

V_i = velocity average in each streamtube fps

ΔT_i = temperature rise in each streamtube $(T_E - T_I)^{\circ}F$.

$\Delta T_{b\ avg}$ = mass flow weighted temperature rise across the burner. $^{\circ}F$

A = area

M_i = $\rho_i\ A\ V_i$ = mass flow rate

Then $$\Delta T_{b\ avg} = \frac{\Sigma M_i \Delta T_i}{\Sigma M_i} = \frac{\Sigma \rho_i V_i \Delta T_i}{\Sigma \rho_i V_i} = \frac{\Sigma X_i V_i \Delta T_i}{\Sigma X_i V_i}$$

APPENDIX C

TOTAL PRESSURE PROBE DESIGN

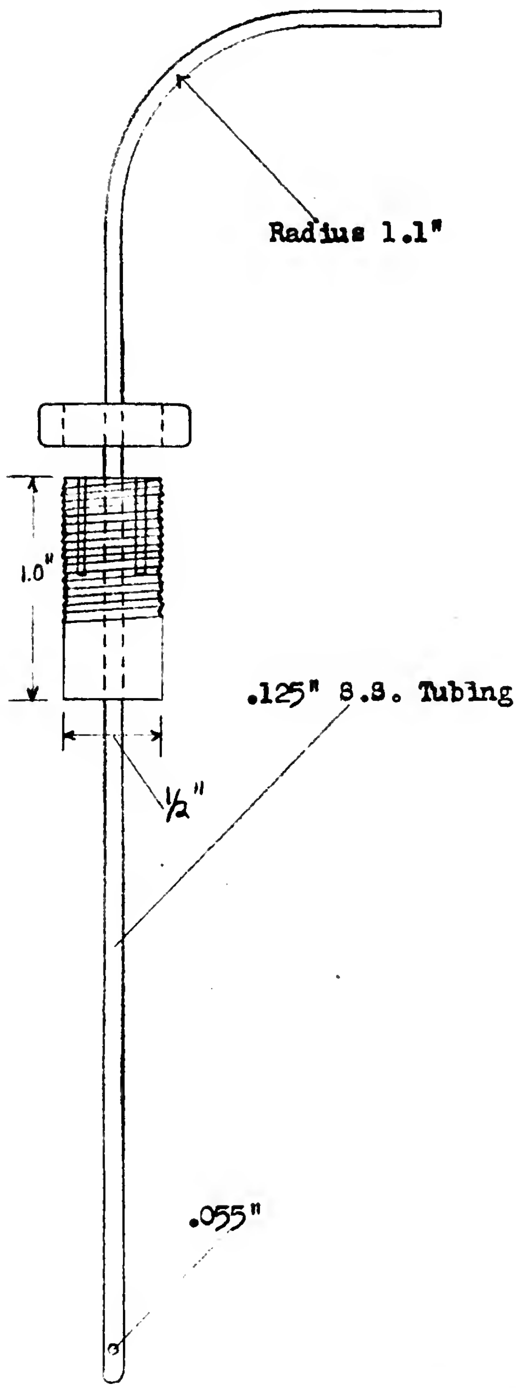
The total or impact pressure can be measured in a fluid stream by inserting a small probe containing an opening whose axis is parallel to the flow direction and then transmitting this pressure to some measuring device. This complies with the general assumption that in subsonic flows the stream is decelerated insentropically to the stagnation point at the hole in the probe and the pressure so measured is the local reservoir or stagnation pressure. Pressure measurements of this type are satisfactory up to the point where local shock waves begin to form around the probe.(1)

The probe was constructed of stainless steel tubing .125 inch in diameter and approximately seven inches in length. A hole .055 inch in diameter was drilled thru one side only, one-fourth of an inch from the bottom. The bottom was sealed by welding. A sleeve was made from mild steel and slotted at the top threaded portion to take a nut for tightening to adjust the friction on the pressure probe. See Fig. 23.

(1) Aerodynamics of a Compressible Fluid, Liepmann and Puckett, John Wiley and Sons, Inc., 1950.

Figure 23

TOTAL PRESSURE PROBE



Scale 1"= 1".

APPENDIX D

FUEL SPLITTER DESIGN

F. C. Mock[1] states that average fuel velocities are 100 fps as they leave the nozzle and that an average included angle of approximately 80° is acceptable. The median full section velocity of 100 fps was chosen as the design point. For this air flow at an air/fuel ratio of 120, the fuel flow was 1.61 cu ft/min = .0268 cfs

$$\text{Vel} = \frac{\text{mass flow}}{\text{area}} = 100 \text{ fps} = \frac{.0268 \text{ cfs}}{\text{area}}$$

$$\text{Area} = .00268 \text{ ft}^2 = .0386 \text{ in}^2$$

The fuel splitter must divide the flow into two equal parts at an angle of 80° and an input velocity of 100 fps. The area above is the total for the flow and .0193 in^2 would be required at the top and bottom of the fuel splitter. The length of the fuel splitter slot is two inches and:

$$A = L \times w \quad \text{or} \quad .0193 \text{ in}^2 = 2 \times w \quad \text{or}$$

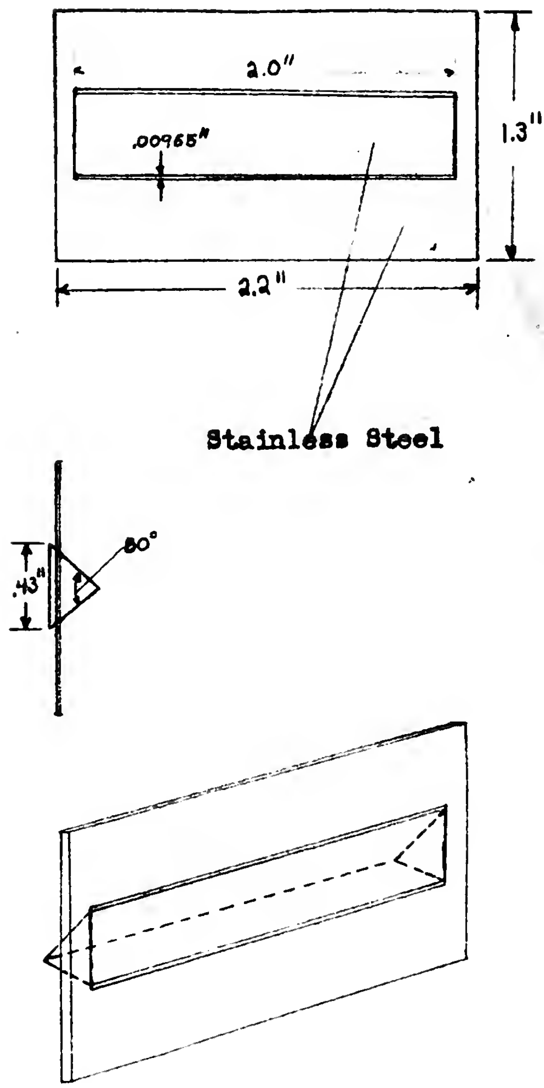
$$W = .00965 \text{ in.}$$

See Fig. 24.

(1) Mock, Frank C., Engineering Development of the Jet-Engine and Gas Burner, SAE Transactions, V. 54, p. 218.

Figure 24

FUEL SPLITTER



Scale 1"= 1".

APPENDIX E

TEMPERATURE PROBE DESIGN

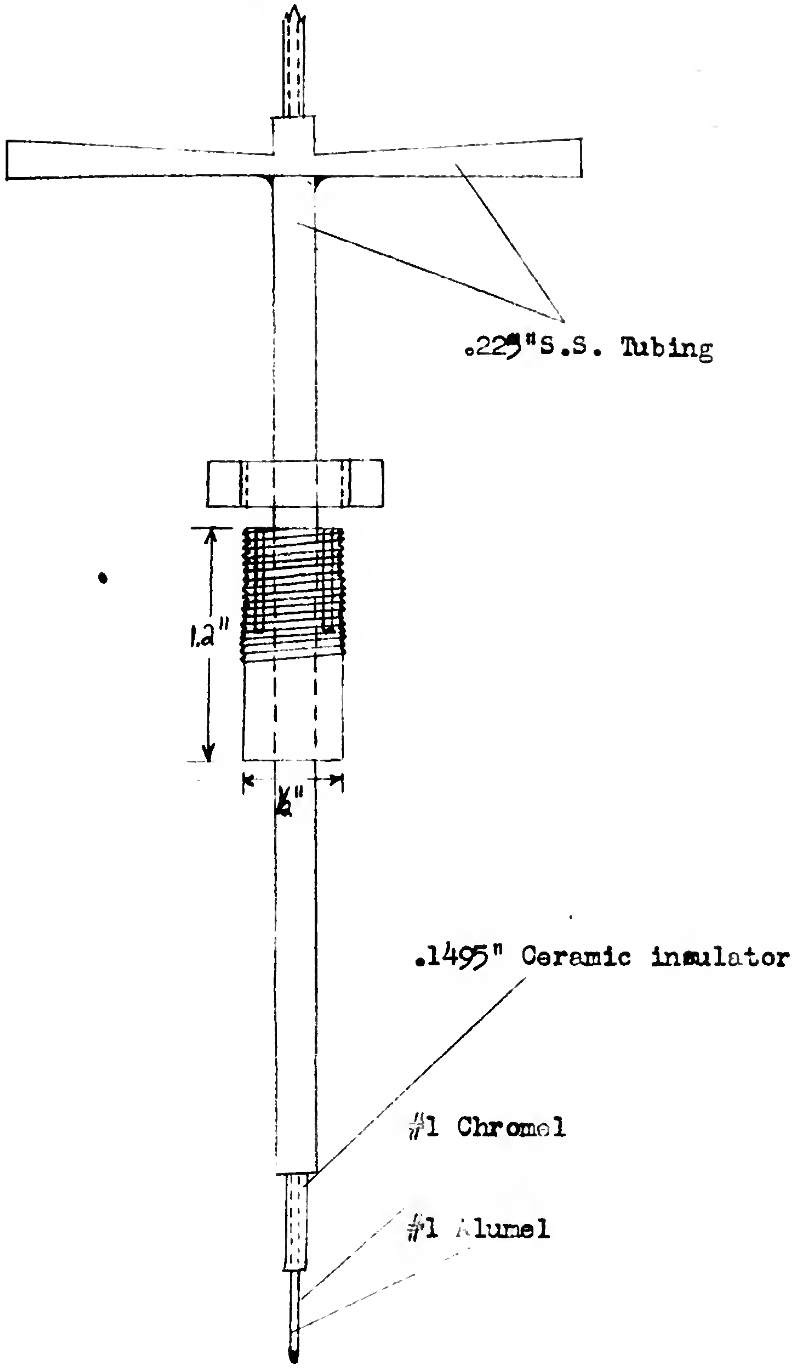
The temperature probe was designed about the principles[1] that bare thermocouples without stagnation attachments are adequate for temperature measurements in air streams at least to 300 fps. At the combustor exit, uneven temperature distributions, large temperature gradients, non-uniform velocity distributions and pronounced temperature fluctuation are sources of error. Bare wire 20 gage chronel-alumel thermocouples although not precise, give satisfactory results.

The temperature probe was designed for and constructed of the same materials as the thermocouples within the combustion chamber. Figure 25 is a full scale drawing of the probe. The body was made from .225 inch stainless steel tubing which was drilled to take .1495 inch diameter double lead ceramic insulators. No. 1 chromel and No. 1 alumel wire was led thru the insulators and welded to form the tip. The tip extended approximately .25 inches below the ceramic insulator and was cemented into permanent place with high temperature ceramic cement.

(1) Hottel, H. C. and Kalitinsky, A., Temperature Measurement in High Velocity Air Streams, Journal of Applied Mechanics, Transactions of the ASME, V. 67, 1945, PA-25.

Figure 25

TEMPERATURE PROBE



Scale 1"= 1".

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cess in a simulated turbo-jet com-
bustion chamber.